

Analysis of Go-Kart Chassis and Optimization of Wheel Hub

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ABSTRACT

The main objective of this paper is to design, to analyze the Go-Kart chassis, and to optimize the wheel hub. Go kart chassis was designed in CATIA software and then analysis was done in ANSYS Workbench. In design methodology, we have discussed how we will formulate our vehicle. DFMEA was carried out then parameters which affect the chassis were studied, chassis was designed keeping in mind various constraints and fitment of components. Maximum stress and total deformation for chassis was calculated using ANSYS workbench. Similarly for brake system braking force and braking torque was calculated which is precisely required to stop the vehicle in motion. Engine for the kart was selected according to the availability and weight distribution was calculated. Topology optimization was carried out to reduce the weight and to reduce the scrub radius of tire which is in contact with the road, the design was optimized in such a way that it also contributes towards ease of manufacturing. After designing, packaging of vehicle was done in CAD software.

Keywords: Go-Kart, DFMEA, Topology optimization.

* Masterminded EasyChair and created the first stable version of this document

* Created the first draft of this document

INTRODUCTION

The chassis forms the central frame of a vehicle, which carries all the part and carries all the loads. These loads include the weight of each parts and the forces such as during acceleration, braking and during cornering. Therefore, the vehicle chassis is considered as the most significant element of the vehicle as it holds all the various parts and vehicle components together [1].

Chassis should be designed well as it is significant to ensure the safety, performance and roadworthiness of the vehicle. The chassis was made of steel tube, there are no suspensions therefore chassis have to be pliable enough to work as a suspension and stiff enough so that it will not break on a turn. The important part of steering is to make sure that the wheels are pointing and can move in the directed directions. This is typically due to series of linkages, rods, pivots and gears [2]. This report explains the methodology to design, fabricate and test the chassis, knuckle, hub of kart. The Design was approached by considering all possible ways for the structure and modeling them in CATIA software and put through the analysis process using ANSYS 16.0 FEA tool. From the

analysis result, the model was altered, retested and a final design was freezed. The formulation process of the vehicle is based on various engineering features depending upon the following factors- Safety and Ergonomics, Availability in market, Costing of the components, Safe engine operations and overall vehicle performance [4]. In this work the basic chassis model has been developed in the CATIA software various design parameters have been considered while designing the chassis like Factor of safety, Ergonomics, adequate bending stiffness, type of chassis to be designed, cost, weight, structural design. Engine used for designing the go-kart is Pulsar in go karts normally only one rear disk brake is present. Steering system is the significant part of any automobile design for the smooth change in directions. For go-kart Ackermann principal of steering is used. It enables smooth and stable maneuvering of the vehicle, the main advantage of Ackerman steering geometry is that it makes the inside wheel steer to a wider angle than the outside wheel. After designing the chassis in CATIA software then the analysis was done in ANSYS software various Impact test was carried out on ANSYS software such as front, rear, and side impact test. Various results have been plotted considering FOS and results were under defined limit [3].

1.1 Materials and methods

At the initial stage review of existing studies was done after that Design Failure Mode and Effect Analysis (DFMEA) of several components were carried out. After determining the RPN number from DFMEA preventive measures for same was mentioned and the aim was to reduce the RPN number.

1.2 Design Failure Mode Effect Analysis (DFMEA).

For designing a kart the first and foremost thing done was to review the case studies going on in Go-kart. Then identification of various components in go-kart was analyzed. Then DFMEA was carried out, it was first done by recognizing all the parts of the go-kart. This is then marked up by observing modes of failure for each part, and the effect that causes on the particular single part and the whole moving kart. In DFMEA for every modes of failure the failure severity, how would be the failure occur, and how will it be detected is determined. The numerical ranking for these parameters is given as 1 to 10. Then these ratings are then multiplied to evaluate the index called RPN. Failure severity, its occurrence and its detection are all product of RPN as shown in equation 1.

$$\text{RPN} = (\text{Severity Rating}) \times (\text{Occurrence rating}) \times (\text{Detection rating}) \dots\dots\dots (1)$$

Techniques of prevention for each modes of failure are recorded after the above step and proper action is taken according to the prime concern of the modes of failure. To minimize the RPN value is the basic focus of the method. DFMEA was executed on 12 main parts of the Go-kart. The analysis was done on the parts namely. Transmission, Frame, pedals, rim, tires, braking system, steering wheels, knuckles, engine, bracket, steering column. The detailed DFMEA process is illustrated in table1.

According to their RPN ranking of parts, according to their risk of failure it was observed that support, frame, steering, knuckles were in top 5. Quality of go-kart is critical when it is observed when their RPN value is above 250. Detailed precaution techniques were noted down and correct actions were taken for these parts as well as other parts of the go-kart.

2 DESIGN OF CHASSIS

The starting stages of chassis design has 4 major cross section namely the roll hoop at front, main roll hoop, and two side members. The wheelbase and track width was finalized then the chassis was designed in CATIA software trial runs were taken in CATIA after 26 trial runs the chassis was finalized for trial runs and analyzing the results ANSYS software was used. The design method of this kart is iterative process and it is based on engineering and reverse engineering process. The entire go-kart design is modeled by keeping in mind that it should be able to withstand racing conditions without and any failure.

The primary purpose of the roll cage is to safeguard the driver and to form a rigid support to mount the components i.e. engine, drive train, and axle. For designing the Roll-cage various accepts such as compact design, light weight, ergonomics, durability, ease while manufacturing, low cost, and light in weight should be considered [5].

Table 1 shows different properties of various steels that have been studied. So by referring the above table in my thesis we have selected AISI 1018 grade steel is referred as low carbon steel having 0.14% carbon this steel is more effective than AISI 1010 and AISI 1015 grade steel and its cost is less than AISI 1020 grade steel and properties such as tensile strength, yield strength, are more as we need rigid and stiff chassis so selecting AISI 1018 will be effective.

Table 1: Properties of Steel

Sr. No.	Properties	AISI 1010	AISI 1015	AISI 1018	AISI 1020
1	Density (gm/cc)	7.87	7.87	7.87	7.87
2	Tensile strength (MPa)	365	385	440	420
3	Yield strength (MPa)	305	325	370	250
4	Modulus of elasticity (GPa)	190-210	190-210	205	205
5	Shear modulus (GPa)	80	80	80	80
6	Poisson ratio	0.27-0.3	0.27-0.3	0.29	0.29
7	Elongation in break (50mm)	20%	18%	15%	15%
8	BHN	105	111	126	121
9	Rockwell hardness	60	64	71	68
10	Thermal conductivity (W/mK)	49.8	51.9	51.9	51.9

2.1 CAD model

Chassis was designed using CATIA V5 software as shown in figure 1. Outer diameter of tube is 25.4 mm and thickness is 2 mm, circular in cross-section. Circular cross-section is used in our design so that to control the difficulties of increase in dimensions and shoot in overall weight, circular cross-section can resist twisting moment and incorporates torsional rigidity.

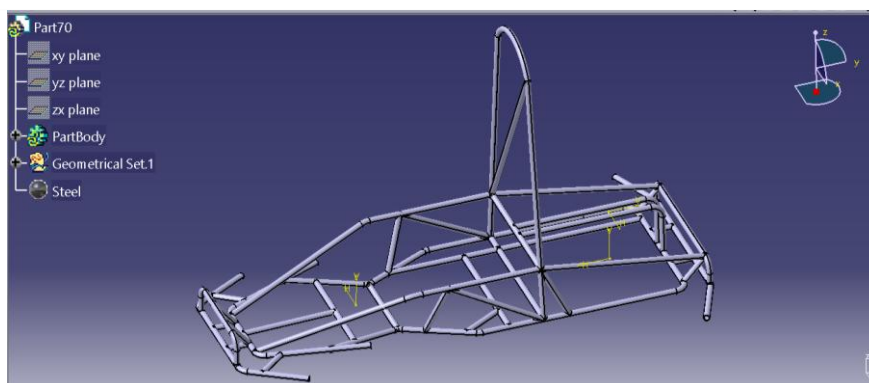


Figure 1: Design of chassis

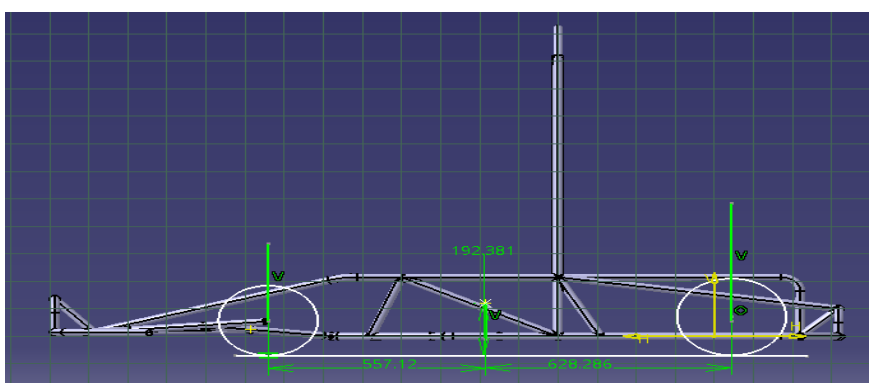


Figure 2: Location of C.G point on chassis

Figure 2 shows the location of C.G point on chassis it is also defined as centre of mass it is that point where all mass is concentrated and all the external forces like acceleration, braking and cornering act through it.

Centre of gravity height, $h = 192.38$ mm.

Location of C.G through front axle, $a = 557.12$ mm.

Location of C.G through rear axle, $b = 628.28$ mm

2.2 Chassis Specification

Table 2 shows the specification of chassis which is calculated from the design mode in CATIA.

Table 2: Chassis specification

Sr. No.	Parameters	Dimensions
1	Wheelbase	1185 mm
2	Front track width	900 mm
3	Rear track width	990 mm
4	Total height of chassis	1144.89 mm
5	Total length of chassis	2035 mm
6	Total width of chassis	1122.19 mm

3 Component Design and selection

i. Engine

Engine used for our thesis was Bajaj Pulsar 150cc DTIS air-cooled engine which approximates the total weight around 5 Kg. Engine is sorted on basis of following parameter -

- (a) Leading OEM in the market. (b) Easy of availability.

Based on the above mentioned parameters as pulsar 150cc engine was effective in delivering torque needed and was also cost effective. Table 3 shows the specification of engine and figure 3 shows the CAD model of engine.

Table 3: Engine specification

Sr. No.	Element Name	Values
1	Engine technology	DTSI, single cylinder 4 stroke, air cooled engine
2	Maximum power	8500 rpm
3	Torque	13.25 Nm
4	Displacement	149.5cc
5	Weight	5 kg
6	Small sprocket diameter	65 mm
7	Large sprocket diameter	128 mm
8	Small sprocket teeth.	14
9	Large sprocket teeth.	32
10	Reduction gear ratio.	1.92
11	Chain Length.	1160 mm

a. Engine calculation

Calculation of gear ratio

Parameter for calculation of gear ratio is taken from table 7

No. of teeth on driving gear $N_1 = 14$ (2)

No. of teeth on driven gear $N_2 = 32$ (3)

$$\text{Gear ratio} = \frac{N_1}{N_2}$$

$$\text{Gear ratio} = \frac{32}{14} \dots\dots\dots (4)$$

$$\text{Gear ratio} = 2.28 \dots\dots\dots (5)$$

b. Weight Distribution calculation

In case of 4 wheeler vehicle it is essential that front or rear wheel should not get lift off the ground while the vehicle takes the turn the condition is satisfied as long as the vertical reaction of the ground on any of the wheel is positive in upward direction. Fig 2 shows the C.G location of the kart.

Mass of kart = 130 Kg

Weight of kart = $130 \times 9.81 = 1275.3 \text{ N} = 1 \text{ G}$.

Wheelbase = 1185mm

Front wheel centre to CG, $a = 557.12$

Rear wheel centre to CG, $b = 628.28$

$$\text{Reaction on front wheel due to weight} = \frac{a}{\text{wheelbase}} \times \text{weight of kart} \dots\dots\dots (6)$$

$$= \frac{557.12}{1185} \times 1275.3 = 600 \text{ N upwards} = 0.47 \text{ G}$$

$$\text{Reaction on rear wheel due to weight} = \frac{b}{\text{wheelbase}} \times \text{weight of kart} = \frac{628.28}{1185} \times 1275.3 \dots\dots (7)$$

$$= 676.15 \text{ N} = 0.53 \text{ G}$$

Weight distribution (F: R) = (47:53)

Static load calculation

$$\text{Static load on front wheel} = (0.47 \times M_k) = (0.47 \times 130) = 61.1 \text{ Kg} = 604.2 \text{ N} \dots\dots\dots (8)$$

$$\text{Static load on rear wheel} = (0.53 \times M_k) = (0.53 \times 130) = 68.9 \text{ Kg} = 675.9 \text{ N} \dots\dots\dots (9)$$

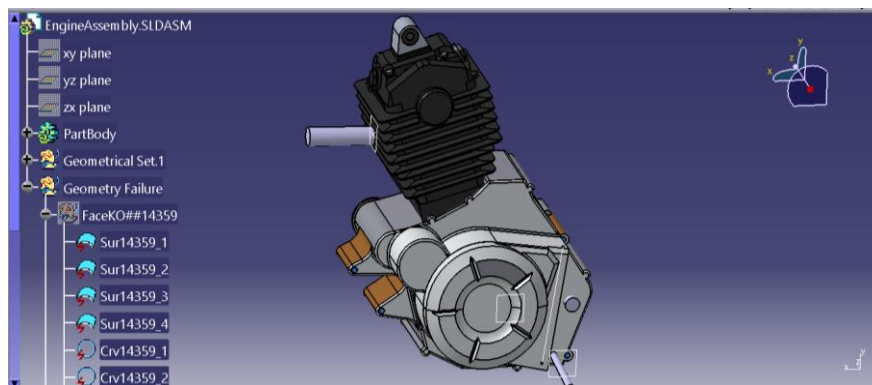


Figure 3:CAD model of engine

ii. Design of Shaft

Material used to manufacture the shaft was AISI 1018, shaft was designed in CATIA software as shown in figure 4

a. Calculation of bending moment

Material chosen is AISI 4140 with

Yield strength = 415 MPa

D = Outer diameter = 40 mm

d = Inner diameter = 37 mm

μ = Coefficient for slicks = 0.9

L = Distance between the wheels and bearings = 170 mm

W = Static weight carried by rear wheel = 675.9 N

Torque = 13250 N-mm

Taking weight distribution = 47:53

$$\text{Bending moment, } M = \sqrt{(M_{\text{weight}})^2 + (M_{\text{friction}})^2} \dots\dots\dots (10)$$

$$M_{\text{weight}} = \mu W = 0.9 \times 675.9 = 608.3 \text{ N-mm}$$

$$M_{\text{friction}} = \mu WL = 0.9 \times 675.9 \times 170 = 103.41 \times 10^3 \text{ N-mm}$$

$$M = \sqrt{(608.3)^2 + (103.41 \times 10^3)^2} = 103.41 \times 10^3 \text{ N-mm}$$

b. Calculation of shear stress

Torque = 13250 N-mm

$$\text{Polar section modulus, } Z_p = \frac{\pi(D^4 - d^4)}{16D} = \frac{\pi(40^4 - 37^4)}{16 \times 40} = 3.36 \times 10^3$$

$$\text{Maximum shear stress, } \tau_{\max} = \frac{\sqrt{M^2 + T^2}}{Z_p} = \frac{\sqrt{(103.41 \times 10)^2 + (13250)^2}}{3.36 \times 10^3} = 3.95 \text{ Mpa}$$

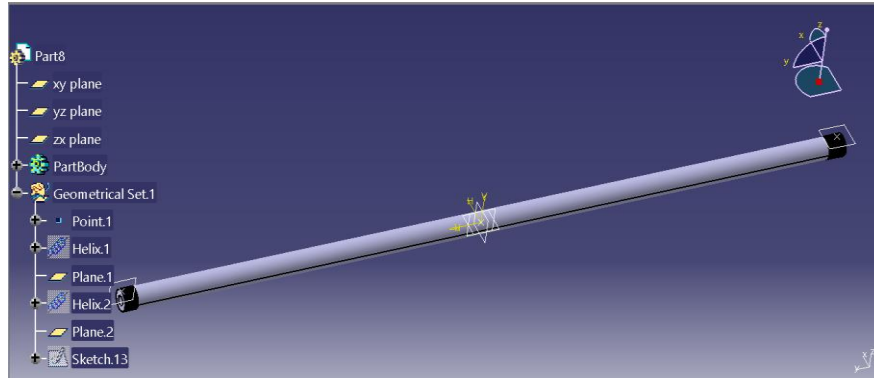


Figure 4: CAD model of shaft

iii. Design of Brakes

In our thesis, we have used moped brake assembly and theoretical calculations were performed. Table 4 shows the braking parameters in which dimensions of caliper and master cylinder were taken and figure 5 shows the CAD model of brake.

Table 4: Braking parameter

Sr. No.	Brake type	Hydraulic disk brake
1	Master cylinder diameter	10 mm
2	Calliper piston diameter	20 mm
3	Pedal ratio	4:1
4	Pedal force applied	20 Kg, 196.2N

Brake calculation

Mass of kart, $M_k = 130 \text{ Kg}$

Weight of Kart, $W_k = 130 \times 9.81 = 1275.3 \text{ N}$

Height of CG, $h = 192.38 \text{ mm}$

Wheelbase, $W_b = 1185 \text{ mm}$

Track width = 989.34mm

Coefficient of friction, brake pad, $\mu_p = 0.4$

Coefficient of friction linking tyre and road, $\mu_t = 0.71$

Weight distribution ratio = 47:53

Static load on front wheel, $S_{if} = 61.1 \text{ Kg} = 604.2 \text{ N}$

Static load on rear wheel, $S_{ir} = 68.9 \text{ Kg} = 675.9 \text{ N}$

$$\text{Weight transfer, } W_t = \mu_r \times C.G \times \frac{M_k}{W_b} = 0.7 \times 192.38 \times \frac{130}{1185} = 14.77 \text{ Kg. (11)}$$

Static Calculations

a) Dynamic front wheel weight transfer during braking

$$D_{wf} = S_{lf} + W_t = 61.1 + 14.77 = 75.87 \text{ Kg}$$

b) Dynamic rear wheel weight transfer during braking.

$$D_{wr} = S_{lr} - W_t = 68.9 - 14.77 = 54.13 \text{ Kg}$$

c) Braking force $B = M_k \times \mu_r \times g = 130 \times 0.7 \times 9.81 = 892.71 \text{ N}$

Required braking force on rear $B_f = D_{wr} \times \mu_r \times g = 54.13 \times 0.7 \times 9.81 = 371.71 \text{ N}$

Required braking force on rear $B_t = B_f \times R_{tyre} = 371.71 \times 0.1397 = 51.92 \text{ Nm}$

d) Disk design calculation

$$\text{Disk brake clamp load, } C_f = \frac{T}{r_e \cdot \mu_p \cdot n}$$

Where, F = force on pad, r = effective mean radius, $\mu_p = 0.4$. n = No. Of friction faces = 2

T = torque on brake = 51.25 Nm, r_e = effective radius, μ_f = coefficient of friction, disk lining material

$$C_f = \frac{51.25}{0.0725 \times 0.4 \times 2} = 883.62 \text{ N}$$

$$\text{Mean effective radius, } E_m = \frac{(D+d)}{4}$$

$$\text{Mean effective radius, } E_m = \frac{(170+120)}{4} = 72.5 \text{ mm}$$

Torque capacity of 2 pads, $T = 2 \times \mu_p \times C_f \times r_e$

$$51.25 = 2 \times 0.4 \times 883.62 \times r_e$$

$$r_e = 0.0725 \text{ m}$$

So effective diameter of disk, $d_e = 0.145 \text{ m} = 145 \text{ mm}$

For safe design and availability we have taken standard disk diameter as $d_e = 170 \text{ mm}$

Practical calculation

Considering pedal force 20 Kg maximum and 10 Kg minimum, so Pedal ration comes to be 4:1

a) Maximum Force on master cylinder piston

$$F_{mc_{max}} = P_{f_{max}} \times g \times 4 = 20 \times 9.81 \times 4 = 784.8 \text{ N}$$

b) Minimum Force on master cylinder piston

$$F_{mc_{min}} = P_{f_{min}} \times g \times 4 = 10 \times 9.81 \times 4 = 392.4 \text{ N}$$

Consider piston of master cylinder, $D_{mc} = 15 \text{ mm} = 0.015 \text{ m}$

$$\text{Area of master cylinder piston, } A_{mc} = \frac{\pi}{4} \times D_{mc}^2 = \frac{\pi}{4} \times 0.015^2 = 1.76 \times 10^{-4} \text{ m}^2$$

$$\text{Maximum Brake line pressure, } BLP_{max} = \frac{F_{mc_{max}}}{A_{mc}} = \frac{784.8}{1.76 \times 10^{-4}} = 4.45 \times 10^6 \text{ N/m}^2$$

$$\text{c) Minimum Brake line pressure, } BLP_{min} = \frac{F_{mc_{min}}}{A_{mc}} = \frac{392.4}{1.76 \times 10^{-4}} = 2.22 \times 10^6 \text{ N/m}^2$$

$$\text{Area of clipper piston, } A_{cp} = \frac{\pi}{4} \times d_{cp}^2 = \frac{\pi}{4} \times 0.020^2 = 3.14 \times 10^{-4} \text{ m}^2$$

d) Maximum Rotating force, $R_{f_{max}} = BLP_{max} \times A_{cp} \times 2 \times \mu_p \times n$

$$= (4.45 \times 10^6) \times (3.14 \times 10^{-4}) \times 0.4 \times 2 \times 2 = 2235.68 \text{ N}$$

e) Minimum Rotating force, $R_{f_{min}} = BLP_{min} \times A_{cp} \times 2 \times \mu_p \times n$

$$R_{fmin} = (2.22 \times 106) \times 3.14 \times 0.4 \times 2 \times 2 = 1.11 \times 10^3 \text{ N}$$

f) Maximum Braking torque, $Bt_{max} = R_{fmax} \times \text{effective mean radius} = 2235.68 \times 0.0725 = 162.08 \text{ Nm}$

g) Minimum Braking torque, $Bt_{min} = R_{fmin} \times \text{effective mean radius} = (1.11 \times 10^3) \times 0.0725 = 80.47 \text{ Nm}$.

h) Maximum Braking force, $B_{fmax} = \frac{Bt_{max}}{R_{tyre}} = \frac{162.08}{0.1397} = 1160.2 \text{ N}$

j) Minimum Braking force, $B_{fmin} = \frac{Bt_{min}}{R_{tyre}} = \frac{80.47}{0.1397} = 576.02 \text{ N}$

As mentioned above required braking torque is 51.25 Nm

The braking torque at min pedal force of 10 Kg is 80.47 Nm

So considering Braking efficiency = 80%.

The resultant braking Torque is $\frac{80.47}{100} \times 80 = 64.37\%$

k) Maximum Deceleration, $D_{max} = \frac{B_{fmax}}{M_k} = \frac{1160.2}{130} = 8.92 \text{ m/s}^2$

Minimum Deceleration, $D_{min} = \frac{B_{fmin}}{M_k} = \frac{576.02}{130} = 4.43 \text{ m/s}^2$

l) Stopping distance

Case 1 :- At speed 40 Kmph, velocity $u = 11.11 \text{ m/s}$

$$S.D = \frac{u^2}{2 \times D_{max}} = \frac{11.11^2}{2 \times 8.92} = 6.90 \text{ m}$$

Case 2:- At speed 50 Kmph, velocity $u = 13.88 \text{ m/s}$

$$S.D = \frac{u^2}{2 \times D_{max}} = \frac{13.88^2}{2 \times 8.92} = 10.79 \text{ m}$$

m) Maximum stopping time

Considering the vehicle running at 40 Kmph

Case 1 :- Considering the vehicle running at 40 Kmph having final velocity (v)

= 0 and initial velocity (u) = 11.11 m/s .

$$v = u + at$$

$$0 = 11.11 - 9.81t$$

$$t = 1.3 \text{ sec.}$$

Case 2:- Considering the vehicle running at 50 Kmph having final velocity (v) =

0 and initial velocity (u) = 13.88 m/s

$$v = u + at$$

$$0 = 13.88 - 9.81t$$

$$t = 1.41 \text{ sec.}$$

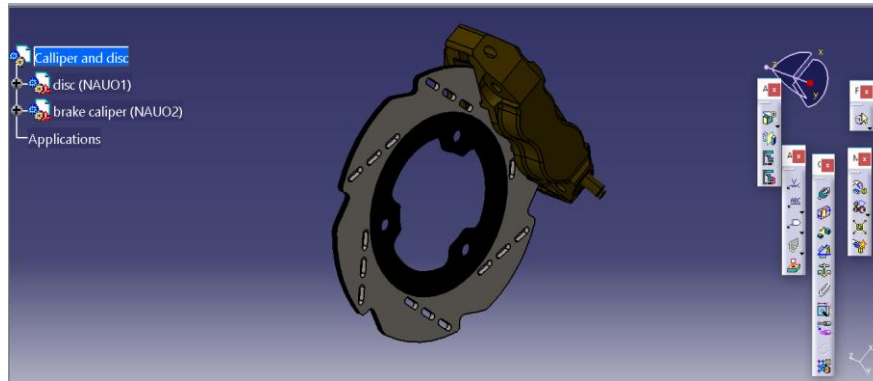


Figure 5: Brakedisc and standard calliper assembly

iv. Design of wheel hub

Wheel hub is situated at the rear and is connected to the rear axle shaft wheel is bolted on it there is roller bearing in between the axle hub and axle shaft which ensures trouble free rotation of wheels. In our thesis we have designed the hub by taking various parameters such as braking force and vertical load that act on the wheel during bump. There are various material available in the market to manufacture the hub but taking the consideration of weight optimization we have manufacture our hub by selecting aluminium as the material. Table 5 shows the specification of materials and figure 6 shows the CAD model of hub.

Table 5: Specification of hub material

Material	Weight	Cost
Steel block	2.18Kg	250/Kg
Aluminium block	1.6Kg	300/Kg

By taking weight and cost under consideration circular aluminium block is selected having diameter 84mm and length of 50mm. Then manufacturing is done on lathe machine.

Calculation of wheel hub

a. Force analysis on wheel hub.

For analysing the forces on hub we consider braking force (F_b) which is transferred to the hub. Equation shows the required braking force.

$$F_b = 892.71 \text{ N} = \text{taking } 1000\text{N for analysis.}$$

b. Static Force (F_{st}) = vertical force acts at the contact of road and tyre when the vehicle comes across the bump this force is greater than the static force.

As front to rear weight distribution is 47:53

$$F_{st} = \frac{m \times 0.53}{2} = \frac{130 \times 0.53}{2} = 34.45\text{N}$$

m = mass of kart

As the vehicle comes across the bump vertical load acting is taken as three times of gravitational acceleration, this force is transmitted to hub centre.

$$F_w = 3 \times g \times F_{st} = 3 \times 9.81 \times 34.45 = 1559.7 \text{ N}$$

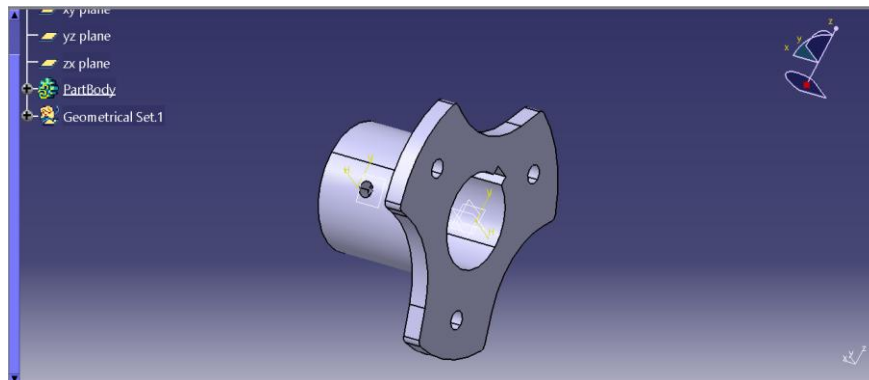


Figure 6:CAD model of optimized wheel hub

4 Assembly of Kart



Figure 7:Kart assembly

5 Results and Discussion

Analysis was been carried out on chassis using ANSYS 16.0, the subsequent calculations were done to calculate impact load.

Considering the outline in which vehicle hits the immobile object with the velocity of 64 Km/hr (17.77 m/s) and the standard impact duration is set to 0.35s. After striking, the final velocity will be 0 m/s and the collision assumed to be elastic in nature.

i. Front Impact Analysis

$$\text{Impact force} = m \times \frac{(V_f - V_i)}{2 \times t_i} = 130 \times \frac{(17.77 - 0)}{2 \times 0.3} = 3850.16\text{N} = 4000\text{N}$$

m= mass of vehicle. Vf = final velocity, Vi = initial velocity, ti = impact time, n = g factor

To calculate G force

$$F = m \times g \times n \quad \dots\dots\dots (12)$$

$$n = \frac{F}{(m \times g)} = \frac{4000}{(130 \times 9081)} = 3.13\text{G} = 4\text{G}$$

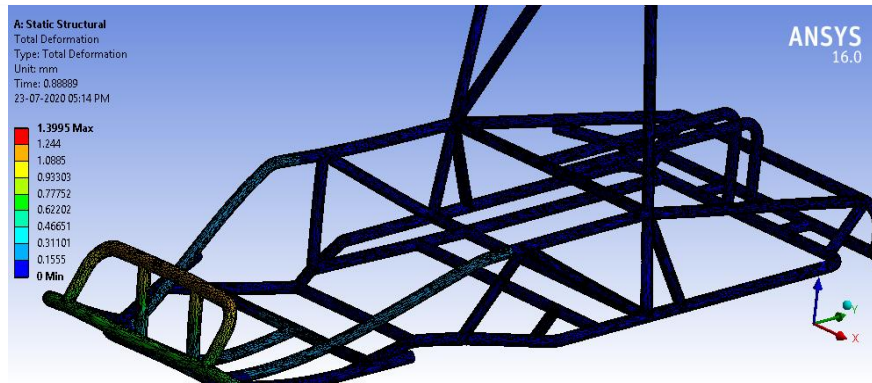


Figure 8: Total Deformation

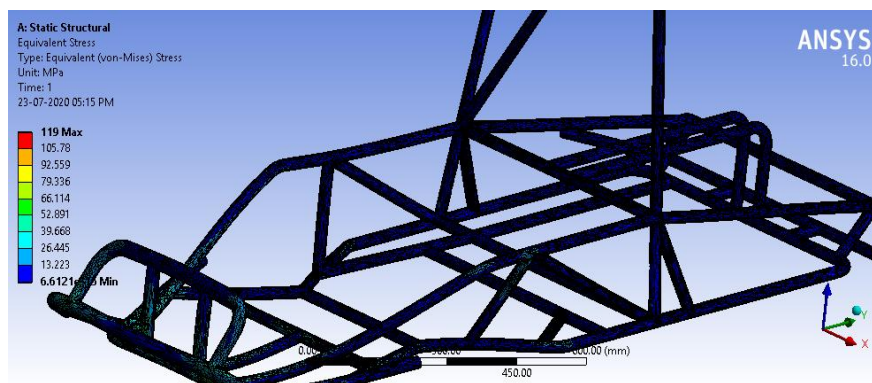


Figure 9: Equivalent Von-Misses Stress.

On applying 4G force, maximum deformation of 1.39 is observed on the chassis figure 8 and the deformation is within the acceptable limit i.e. 2 mm and figure 9 shows the maximum stress is found to be 119 Mpa.

Calculating Factor of safety:-

Working stress = 119 Mpa.

Yield strength of AISI 1018 = 370 Mpa.

$$FOS = \frac{\text{Yield stress}}{\text{Working stress}} = \frac{370}{119} = 3.10$$

ii. Rear Impact Analysis

Once the CAD geometry is imported in ANSYS workbench the skeleton is meshed with fast transition and course relevance centre.

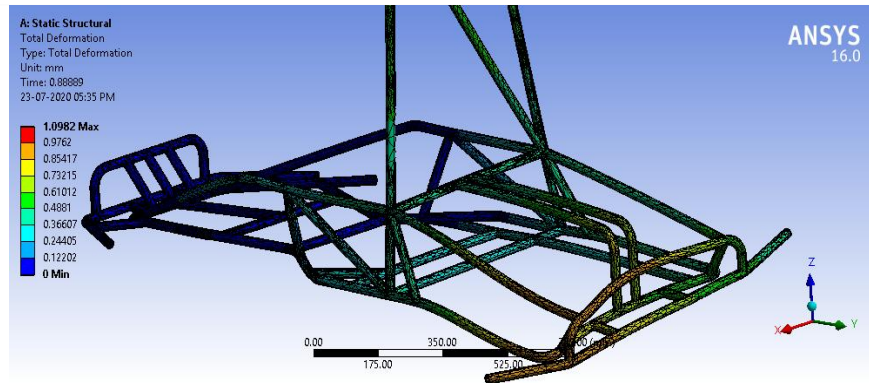


Figure 10: Total Deformation

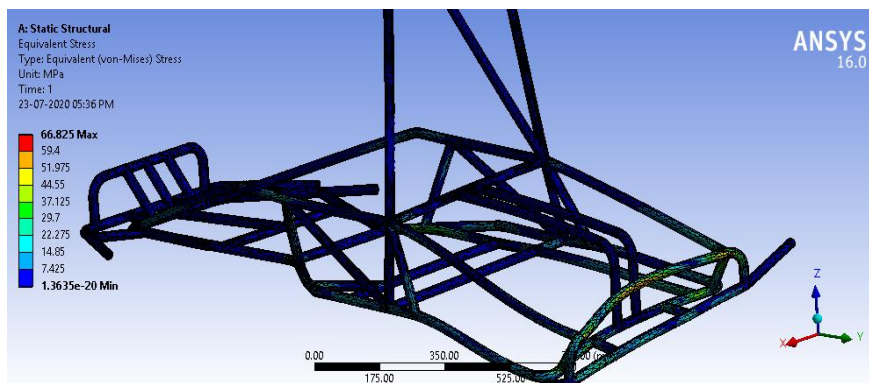


Figure 11: Equivalent Von-Mises Stress

On applying 4G force, maximum deformation of 1.098 is noticed on the chassis figure 10 and the deformation is within the acceptable limit i.e. 2 mm and figure 11 shows the maximum stress is found to be 66.82 Mpa.

Calculating Factor of safety:-

Working stress = 66.82 Mpa.

Yield strength of AISI 1018 = 370 Mpa.

$$FOS = \frac{\text{Yield stress}}{\text{Working stress}} = \frac{370}{66.82} = 5.53$$

iii. Side Impact Analysis

For side impact analysis considering 3G force and assuming time of impact equals to 0.45s.

$$\text{Impact force} = m \times \frac{(V_f - V_i)}{2 \times t_i} = m \times \frac{(17.77 - 0)}{2 \times 0.4} = 2887.62 \text{ N} = 3000 \text{ N.}$$

Calculating G force

$$n = \frac{F}{(m \times g)} = \frac{3000}{(130 \times 9.81)} = 2.26 = 3$$

Once the CAD geometry is imported in ANSYS workbench the skeleton is meshed with fast transition and course relevance centre.

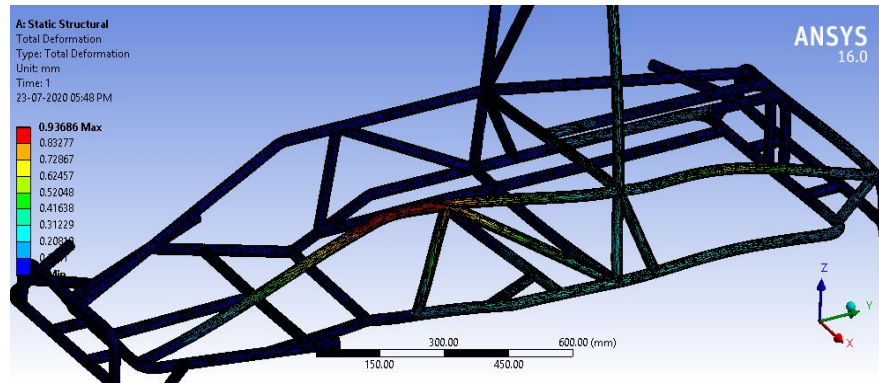


Figure 12: Total Deformation

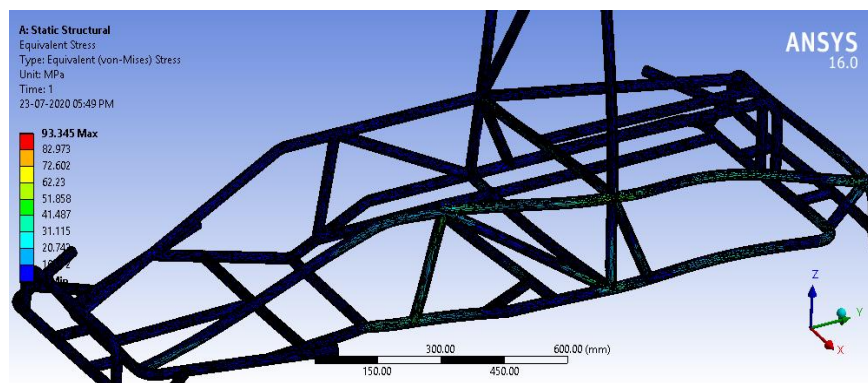


Figure 13: Equivalent Von-Mises Stress

On applying 3G force, maximum deformation of 0.936 is noticed on the chassis as shown in figure 12 and the deformation is within the acceptable limit i.e. 2mm and figure 13 shows the maximum stress is found to be 93.34 Mpa.

Calculating Factor of safety:-

Working stress = 93.34 Mpa.

Yield strength of AISI 1018 = 370 Mpa.

$$FOS = \frac{\text{Yield stress}}{\text{Working stress}} = \frac{370}{93.34} = 3.95.$$

6 Topology Optimization of Wheel hub

Figure 14 Shows wheel hub which was designed in CATIA V5 software the material used for hub is Aluminium 6061

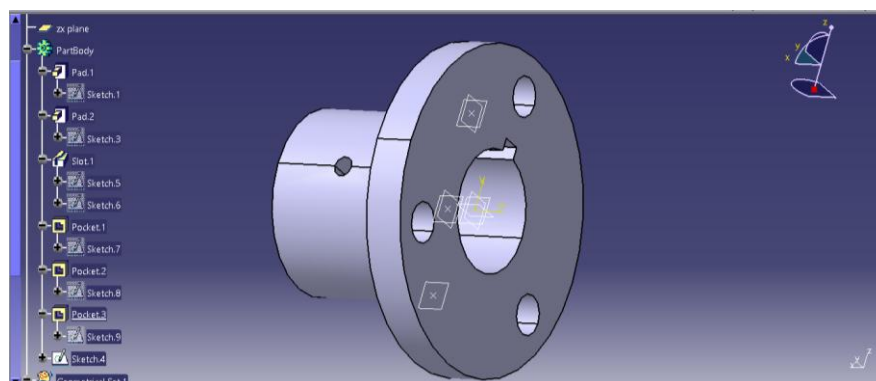


Figure 14: Wheel hub

6.1 Analytical results of old hub

Figure 16 shows the application of bearing loads taking braking force as 1000 N and vertical load as 1559.7 N

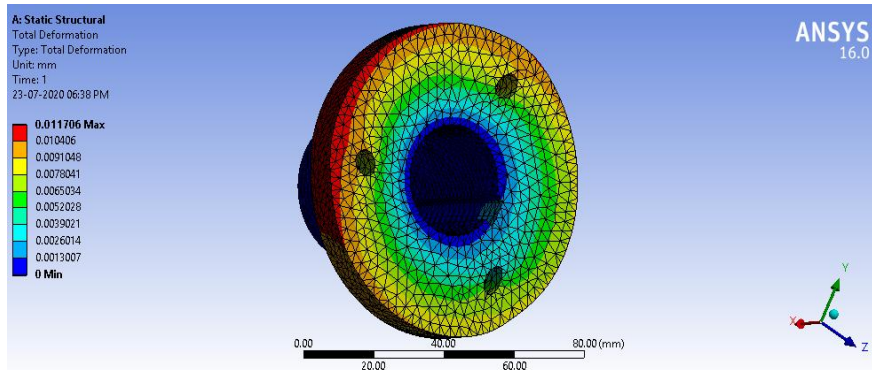


Figure 15: Total Deformation

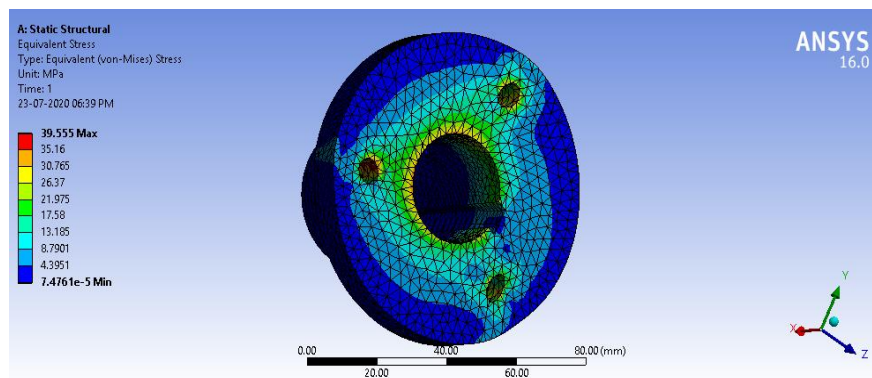


Figure 16: Equivalent Stress

As figure 15 shows the deformation is found to be 0.011mm and figure 16 shows Equivalent Stress equals to 39.55 Mpa.

Calculating factor of safety:-

Working stress = 39.55 Mpa.

Yield strength of AISI 1018 = 270 Mpa.

$$FOS = \frac{\text{Yield stress}}{\text{Working stress}} = \frac{270}{39.55} = 6.82$$

6.2 CAD model of optimized Hub

Figure 17 Shows optimised wheel hub which was designed in CATIA V5 software the material used for hub is Aluminium 6061

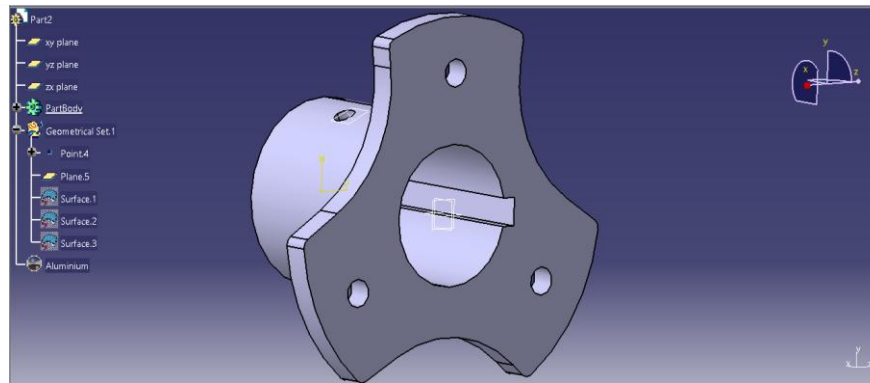


Figure 17: Optimized wheel hub

6.3 Analysis of Optimized wheel hub

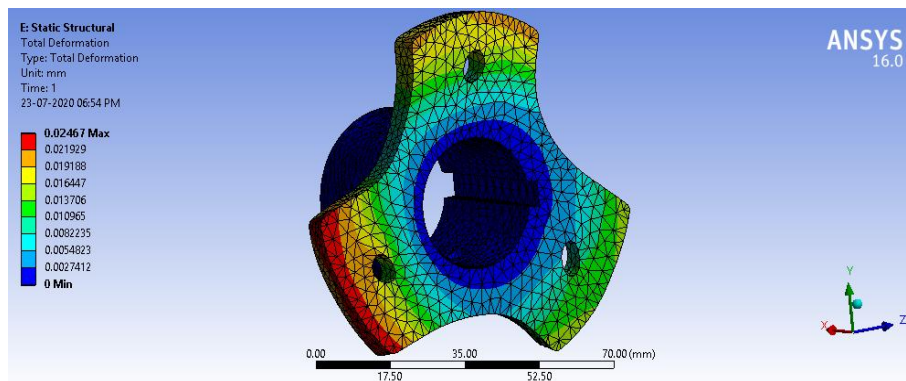


Figure 18: Total Deformation

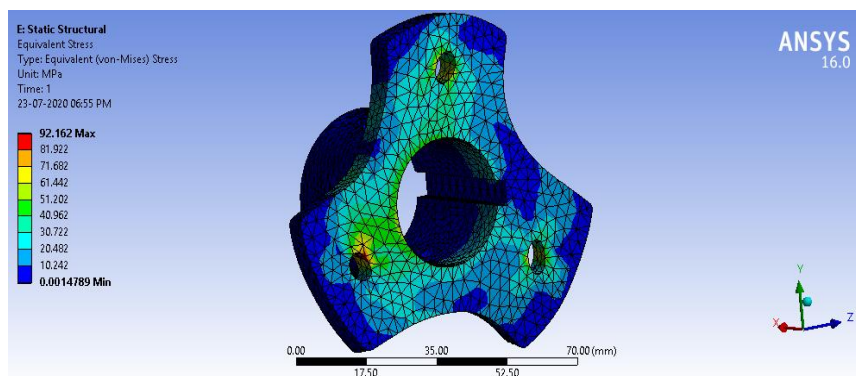


Figure 19: Equivalent Stress

As figure 18 shows the deformation is found to be 0.024mm and figure 19 shows Equivalent Stress equals to 92.16 Mpa.

Calculating factor of safety:-

Working stress = 92.16 Mpa.

Yield strength of AISI 1018 = 270 Mpa.

$$FOS = \frac{\text{Yield stress}}{\text{Working stress}} = \frac{270}{92.16} = 2.92$$

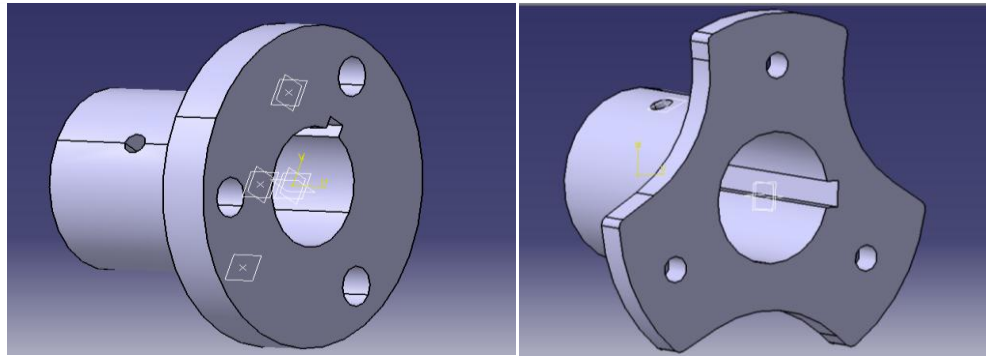


Figure 20:Old Wheel Hub**Figure 21: Optimized Wheel Hub**

Figure 20 shows the old wheel hub and figure 21 shows the optimized wheel hub and the results are plotted in table 6

Table 6: Comparison of Optimized Wheel Hub with the Original Wheel Hub

Sr. no	Parameter	Old wheel hub	Optimized wheel hub	Change	% Change
1	Mass	1 Kg.	0.10 Kg.	0.9 Kg	90%
2	Stress	39.55 Mpa.	92.16 Mpa.	-52.16 Mpa	-133.02%
3	Deformation	0.011mm.	0.024mm.	-0.01mm	-118.18%
4	Factor of safety	6.82.	2.92	3.9	57.18%

The new optimized design of wheel hub is validated with original load conditions and constraints same as that of old one and it is found to be safe.

7 CONCLUSIONS

We have used finite element system to design and evaluate the chassis and other components such as shaft, brake and hub. Our main aim was to reduce the weight of the chassis, to simplify the overall design, and to optimize the wheel hub without sacrificing the durability of kart and the performance. This paper provides adequate knowledge for designing the Go-Kart. Thus after all the theoretical and analytical calculation, it is concluded that this Go-kart is safe under all conditions and meets the performance targets.

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